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335 - Energy And Exergy Analysis Of Advanced Cycles For Solar Cooling

L. Marletta, G. Evola* and F. Sicurella[1]

[1] Department of Industrial Engineering, University of Catania
Viale A. Doria, 6 – 95125 Catania (Italy)

* Corresponding Author, gevola@unict.it

Abstract

In conventional desiccant and evaporative air-conditioning systems (DEC) employing solid sorptive materials, the regeneration stage is powered by thermal energy. The use of solar energy to assist the regeneration process is obviously a good solution to reduce the primary energy consumption. When looking at the problem from a second law perspective, namely by assessing the overall irreversibility production and the exergy efficiency, this practice is not always sufficient to ensure better performance than a conventional air-conditioning system, especially when high regeneration temperatures are required.

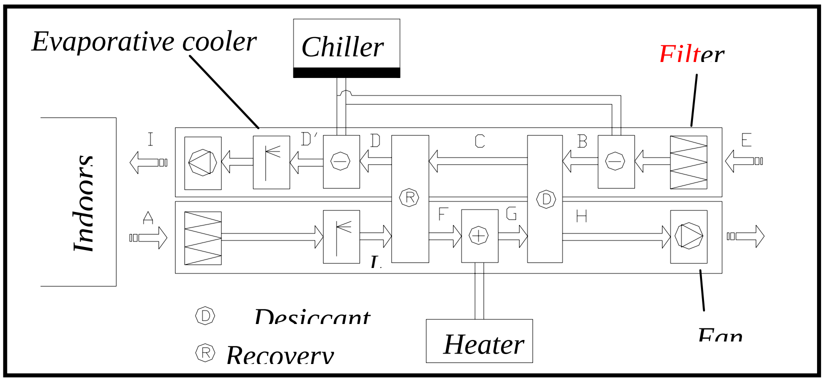
In the literature, new concepts for open cycle desiccant cooling systems have been recently proposed, based on the adoption of enthalpy recovery wheels or heating coils working as a desuperheater on the cooling cycle. In this paper a comparative analysis between the conventional DEC systems and these new concepts is carried out both under a first and a second law perspective, in order to understand to what extent the novel thermal cycles may help reducing thermodynamic irrationalities and irreversibilities, taking into account the possibility of assisting regeneration by means of solar energy. The effects of the parasitic consumption of fans and pumps will be accounted for, too. The results will help understanding possible inefficiencies and improvements of the thermal cycles mentioned above, and highlight the fundamental role of the solar energy.

Keywords: desiccant wheel, solar cooling, exergy, second law performance,

1. Introduction

An air-conditioning system based on the use of desiccant wheels requires thermal energy for the reactivation of the desiccant after the dehumidification process; to this aim the wheel is crossed by a hot air flow, whose temperature depends on the materials and the operating conditions of the system. Thanks to new materials, the regeneration temperatures are around 70 °C, thus the use of solar energy may be suitable and produce relevant energy savings.

Fig. 1 shows an air-conditioning system provided with a desiccant wheel for dehumidification, according to the most common and performing scheme, together with the thermal cycle represented on a psychrometric chart. The air to be supplied to the conditioned space is taken from outdoors, whereas the regeneration flow is taken from the space to be conditioned. The pre-cooling process (EB) is necessary in hot and humid climates, where the only desiccant wheel may not be able to reach the desired inlet humidity ratio. More details concerning the psychrometric processes involved in the system can be found in [1, 2, 3].



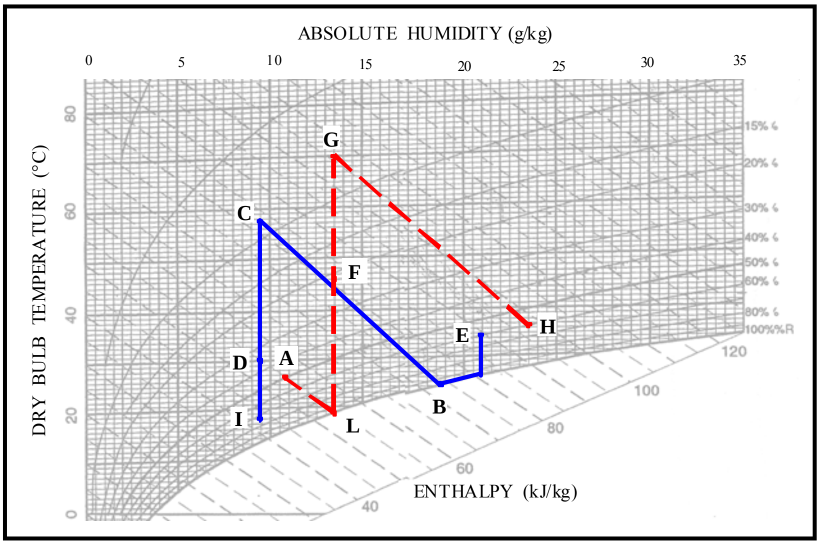


Fig. 1. Lay-out and thermal cycle of a standard DEC system

2. Standard desiccant systems

2.1. The case study

The main purpose of this study is the comparison between the performance of different concepts for DEC systems in Mediterranean climates. This comparison will be performed on a first and second law perspective; the analysis will be applied to the following systems:

- Conventional AHU, whose layout and thermal cycle are shown in [1].
- Standard open-cycle desiccant system (DEC), according to the layout shown in Figure 1.
- Standard DEC provided with a desuperheater.
- Standard DEC modified with the use of an enthalpy recovery wheel, which operates a pre-dehumidification of the process air by releasing a certain fraction of its vapour content to the exhaust air.

For all the thermal cycles related to the DEC systems, the use of solar energy to assist regeneration will also be investigated (solar assisted DEC): the AHU is the same as in the previous points, but a solar section with a back-up heater is adopted to produce hot water for regeneration purposes. In order to test the sensitivity of the results to the size of the solar section, different values of the collector surface will be considered, corresponding to an annual solar fraction F ranging from 0 to 1. In order to compare the energy and exergy performance of the systems, we have considered a case study represented by the ventilation and the air-conditioning of an enclosed space with a latent

thermal load Q_L = 2.5 kW and a sensible thermal load Q_S = 7.5 kW. In Table 1 the design indoor and outdoor conditions are reported, as well as the inlet air conditions, determined by assuming an inlet air flow
m_a = 1 kg/s ($V_a \approx$ 3200 m^3/h); the regeneration air flow is equal to the process air. The design outdoor conditions are typical of a hot and humid climate, such as the one occurring in the countries of the Mediterranean area.

Table 1. Design conditions for the case study.

	Indoor conditions (A)	Outdoor conditions (E)	Inlet conditions (I)
Dry bulb temperature	26 (°C)	35 (°C)	**18.5 (°C)**
Relative humidity	50 %	60 %	**72 %**
Vapour content	10.5 (g/kg)	21.4 (g/kg)	**9.5 (g/kg)**
Enthalpy	**52.8 (kJ/kg)**	**89.9 (kJ/kg)**	**42.6 (kJ/kg)**

2.2. Standard DEC performance

In a previous paper [4], the energy and exergy performance of a solar assisted standard DEC system were studied. In this work some improvements are made to the results of the previous paper:

- The parasitic electric consumption for the operation of fans and pumps are accounted for.
- A new definition for the exergy input to the systems is considered, which accounts for the exergy content of the primary sources, namely the fuel used to feed heaters and power plants for the production of electric energy.

As far as the first point is concerned, the pressure losses Δp_a occurring in each component of the air- handling unit were determined by means of data provided by the manufacturers, as reported in Table 2; an additional pressure loss of 150 Pa was adopted to account for the air distribution system, which may be considered the same for all the systems. The pressure losses Δp_w inside the pipes for the distribution of hot and cold water were also appropriately assessed by taking into account the volumetric water flow rate (V_w), the pipe diameter and its length. The parasitic electric consumption for fans and pumps can then be assessed, assuming η_f = 0.7 and η_p = 0.6 as the efficiency of fans and pumps, respectively, and by means of the following formulas:

$$P_{el,fans} = \frac{V_a \cdot \Delta p_{a,tot}}{\eta_f} \qquad P_{el,pumps} = V_w \cdot \frac{\Delta p_{w,tot}}{\eta_p} \qquad \textbf{(1)}$$

As regards the definition of the exergy input to the systems, in this paper the exergy of the primary sources is accounted for, namely the exergy of the fuel burned to feed the heat generator for the production of hot water and the power plants for the production of electric energy. According to this approach, the primary exergy associated with a certain amount of electric power P_{el} and thermal power Q_{hg} may be respectively assessed as:

$$E_P = \frac{P_{el}}{\eta_{el}} \cdot \frac{HHV}{LHV} \qquad E_Q = \frac{Q_{hg}}{\eta_{hg}} \cdot \frac{HHV}{LHV} \qquad (2)$$

Table 2. Pressure losses inside the air-handling units.

Component	Convent. HVAC	Standard DEC		Standard DEC + condensation		Standard DEC + enthalpy wheel	
		Process	*Regen.*	*Process*	*Regen.*	*Process*	***Regen.***
Heating coil	15 Pa	------	20 Pa	------	20 Pa	------	**20 Pa**
Cooling coil	200 Pa	50 Pa (× 2)	------	50 Pa (× 2)	------	50 Pa (× 2)	------
Desiccant wheel	------	250 Pa	250 Pa	250 Pa	250 Pa	250 Pa	**250 Pa**
Recovery wheel	------	200 Pa	200 Pa	200 Pa	200 Pa	200 Pa	**200 Pa**
Enthalpy wheel	------	------	------	------	------	200 Pa	**200 Pa**
Condensation coil	------	------	------	------	40 Pa	------	------
Evaporator	------	100 Pa	100 Pa	100 Pa	100 Pa	100 Pa	**100 Pa**
Filter	**150 Pa**	**150 Pa**	**150 Pa**	**150 Pa**	**150 Pa**	**150 Pa**	**150 Pa**

Here the ratio between the Higher Heating Value (HHV) of the fuel and its Lower Heating Value (LHV) is used because the exergy content of the fuel is related to its HHV, while the useful energy produced during the burning process is proportional to the LHV. In this analysis, natural gas was considered as the fuel used for the heat generator (HHV / LHV = 1.1), whereas oil was adopted for the power plants (HHV / LHV = 1.06). Furthermore, $\eta_{el} = 0.37$ was adopted as the average efficiency for the production of the electric energy and its distribution to the final user, whereas $\eta_g = 0.90$ was used as the efficiency of the heat generator. In Fig. 2 the comparison between conventional HVAC and standard DEC is reported; the comparison is based on three parameters, namely:

- the Specific Primary Energy Consumption (SPEC), defined as the ratio of the Primary Energy Consumption of the system to the overall thermal building load (QL + QS);
- the Specific Irreversibility Production (SPIR), defined as the ratio of the total Irreversibility Production of the system to the overall thermal building load (QL + QS);
- the exergy efficiency ζ, defined as the ratio of the useful exergy output to the overall exergy input, accounted as primary exergy (see Eqn. 3).

$$\zeta = \frac{m_a \cdot e_I}{E_P + E_Q \cdot (1 - F) + Q_{hc} \cdot F \cdot \left(1 - \frac{T_0}{T_{abs}}\right) + m_a \cdot e_A} \qquad (3)$$

In Eqn. 3 we have considered that, when solar energy is used to assist the regeneration process, only a fraction (1-F) of the thermal load Q_{hc} on the heating coil is covered by using fuel; the remaining fraction F is covered by solar energy, whose exergy content is considered proportional to the Carnot factor associated with the absorber plate temperature T_{abs}. The exergy flows and the irreversibility produced by each process are determined by means of the Gouy-Stodola equation, customized for the analysis of HVAC systems and humid air streams [5, 6]; the dead state corresponds to the outdoor conditions.

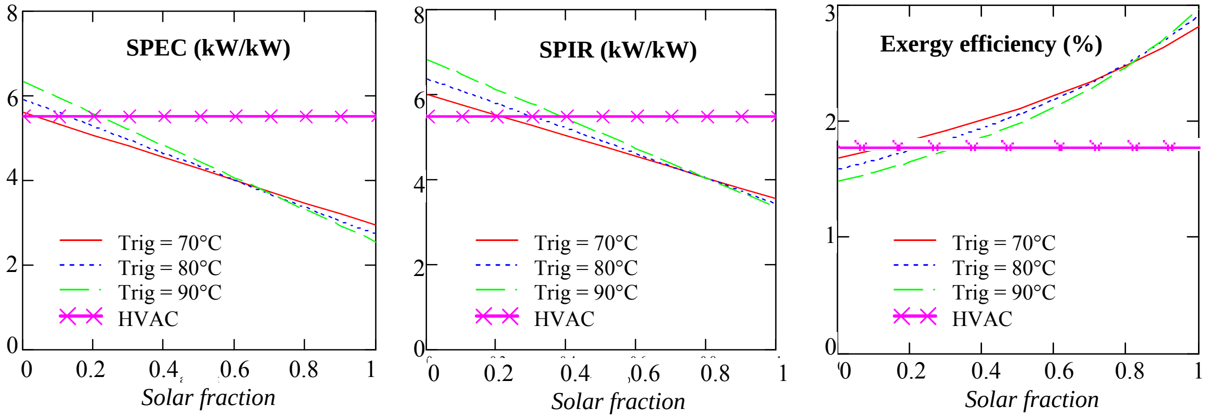


Fig. 2. Energy and exergy performance of conventional HVAC and standard DEC systems.

The standard DEC cycle has been studied for different regeneration temperatures. Thanks to the results shown in Fig. 2, it is possible to define the minimum solar fraction F needed to attain a second law efficiency of the DEC system better than that of a conventional HVAC system. This minimum value depends on the regeneration temperature required by the desiccant wheel; as an example, when working with a regeneration temperature as high as 90°C, at least F = 0.35 is required to get a higher exergy efficiency than the conventional HVAC, whereas F = 0.15 is sufficient if the regeneration temperature is as high as 70°C. When using high regeneration temperatures, a higher thermal power is required by the heating coil, but the dehumidification potential of the desiccant wheel also gets higher, thus allowing a reduction of the size of the pre-cooling coil (EB, see Fig. 1). For this reason, if a very high solar fraction is adopted – more or less higher than 80% – it seems to be more performing to work with higher regeneration temperatures. Further aspects will be addressed later in the paper.
When looking at the first law performance, the advantage of using solar assisted DEC system is more evident, and the energy break-even point is reached for a very low solar fraction. It has also to be underlined that the contribution of the parasitic consumption is not negligible. The exergy provided to feed fans and pumps ranges from 13% of the overall exergy input in a standard DEC system to 21% in a solar-assisted system with F = 1. In the conventional HVAC system this contribution reduces to the 5%, due to the low number of components inside the AHU.

3. Improvements on the standard DEC cycle.

3.1. The adoption of the condensation coil

An improvement of the energy and exergy performance of a standard DEC system may be achieved if an additional heating coil is installed on the regeneration side, after the recovery wheel and before the traditional heating coil; in this paper it will be referred to as “condensation coil”. The amount of thermal energy recoverable from the “condensation coil” is limited by the temperature of the gas coming out of the compressor, which can be assumed as high as 75°C, in a chiller working with R-407C and a condensation pressure around 2 MPa. The saturation temperature of R-407C in the same operating conditions is as high as 45°C, but this potential thermal drop of 30°C can not be always exploited, as the temperature of the air coming out of the recovery wheel can be higher than 45°C, as shown in Tab.3.

Table 3. Inlet air temperature to the condensation coil (point F, see Fig.1)

T_{rig} = 70°C	T_{rig} = 80°C	T_{rig} = 90°C
46,1 (°C)	**51,4 (°C)**	**56,1 (°C)**

Fig.3 shows the results of the analysis as the percentage improvement achievable by using a condensation coil with respect to the corresponding standard DEC cycle. The results get better as both the regeneration temperature and the solar fraction decrease, as in this case the temperature of point C (see Fig. 1) increases, thus determining an increase of the temperature t_F (see Table 3). As previously explained, this reduces the recovery potential of the condensation coil, which causes the limitation of the performance improvement to only 5% if solar energy is not used, against the 12% obtained with a regeneration temperature as high as 70°C.
Furthermore, the advantage of installing a condensation coil in a solar assisted DEC system seems not to be high, as the coil limits the potential left to the solar system.

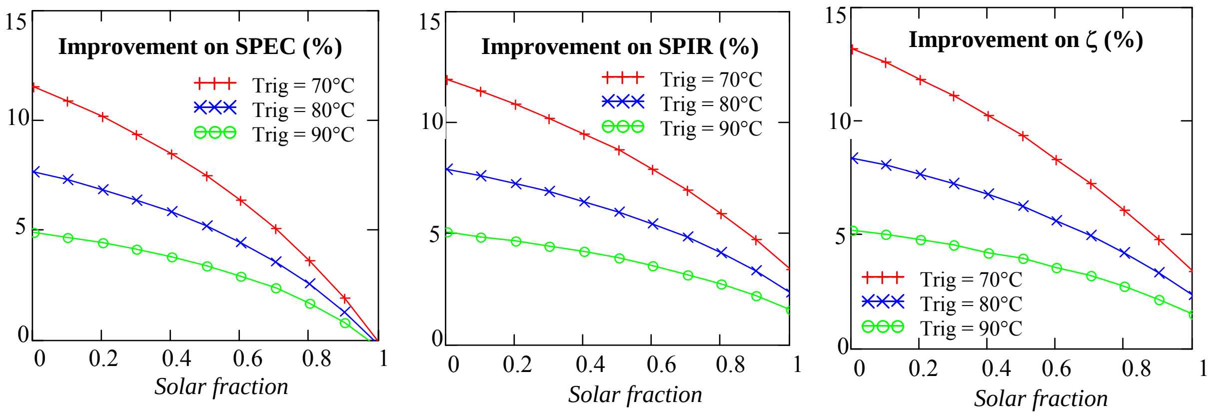


Fig. 3. Improvements due to the adoption of a condensation coil

Actually, there is a different way of looking at these results, which is that the condensation coil allows a reduction of the collecting surface needed to achieve the same results in terms of overall energy savings; an economic analysis is required to decide whether to integrate the system with solar section.

3.2. The adoption of the enthalpy wheel

Another interesting alternative to the standard DEC system is shown in Fig. 4. In this configuration, dehumidification is performed in two different stages; the first stage is carried out by means of an enthalpy wheel, which is similar to the recovery wheel, but in which latent heat can also be exchanged, thanks to the adsorbent materials which cover the rotor surface. Thanks to the enthalpy wheel, a certain amount of water vapour is transferred from the process air to the exhaust air, thus avoiding the use of a pre-cooling coil; but the most important consequence is the need of a lower regeneration temperature, which in this case reduces to 61 °C, given the same inlet conditions for the supply air (point I in Fig. 4). This means that both the cooling and the heating power required by the air-handling unit reduce, allowing important improvements on energy savings and exergy performance, as shown in Fig. 5, where the comparison is made with the standard DEC system working with T_{reg} = 70°C.\

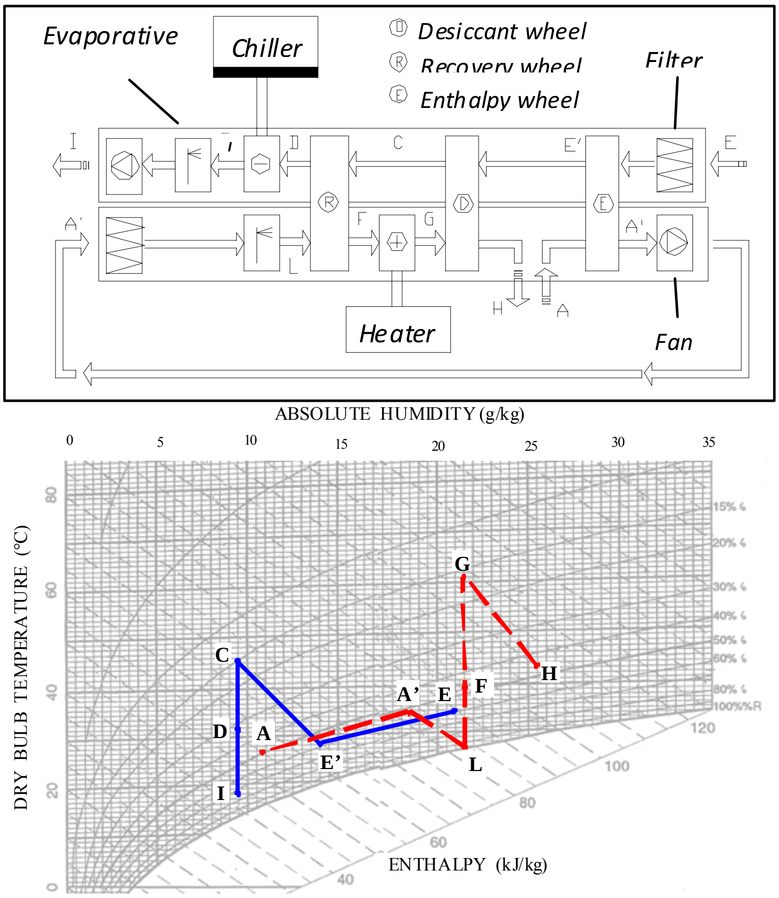


Fig. 4. Lay-out and thermal cycle of a DEC system with enthalpy wheel.

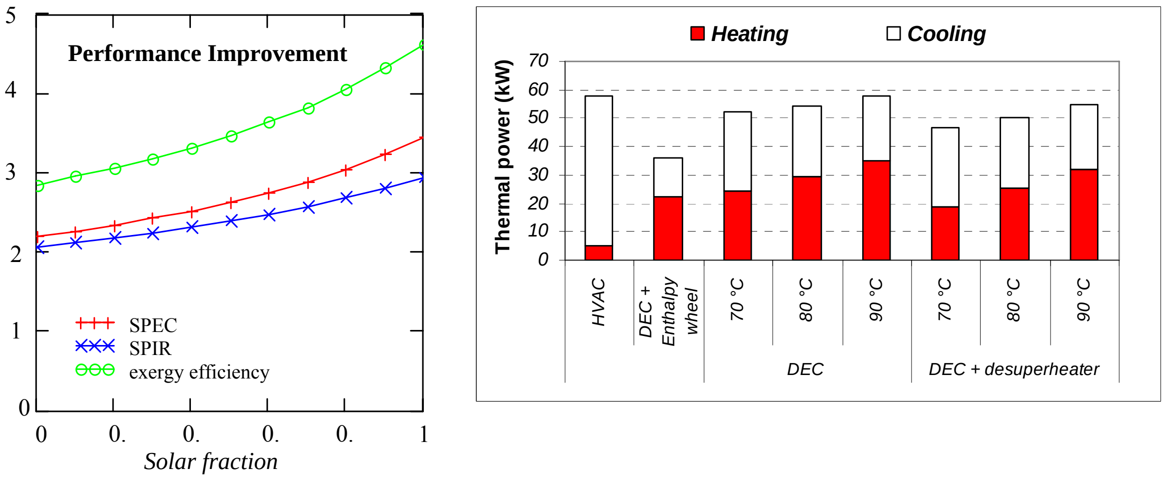


Fig. 5. Improvements due to the adoption of an enthalpy wheel

The most relevant results refer to the exergy efficiency, and are mainly due to the consistent reduction of the cooling power, which is produced through the use of electric energy, characterized by very high exergy content. The adoption of this solution also emphasizes the advantage of using solar energy to assist regeneration, as the performance improvement gets higher for high solar fractions.

4. Conclusions

As shown in the previous paragraphs, it is possible to improve the energy and exergy performance of a standard DEC system by adopting some additional components which allow relevant energy recovery.

Fig.6 summarizes the most important results, and underlines the great advantage of using the solution with an enthalpy recovery wheel. For a right interpretation of the results it is necessary to look at Fig. 7, as a certain solar fraction corresponds to different collecting surface according to the system which is considered; indeed, the entity of the thermal power required by the heating coil depends on the plant scheme. This is an aspect which can not be neglected, especially if the first and second law results of this study must be translated in economic figures. In terms of collecting surface, the results are shown in Fig. 7; they refer to vacuum tube solar collectors and an overall irradiation as high as 800 W/m^2.

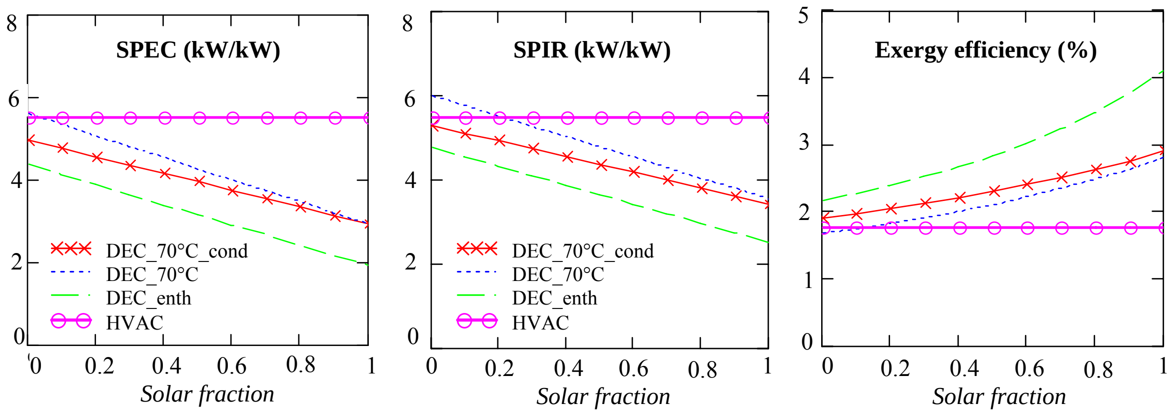


Fig. 6. Summary of the most relevant results of the study.

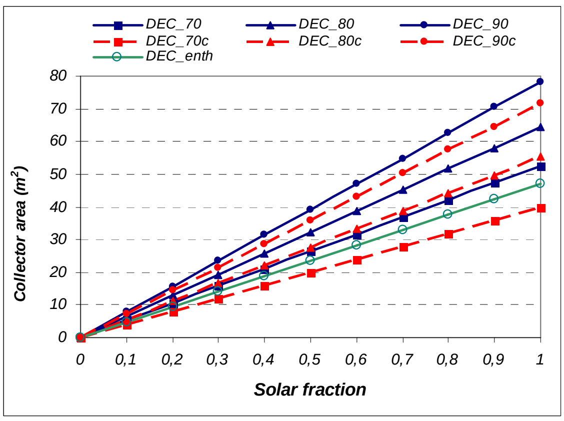


Fig. 7. Solar collecting area as a function of the solar fraction.

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